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by Fred L. Robson and Winthrop E. Hilding

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University of Connecticut

ABSTRACT

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This paper presents a method of predicting the static pressure gradient of a saturated vapor flowing in a round, straight tube with an arbitrary heat flux. The method is based upon a Lockhart-Martinelli type correlation using, however, experimental data for condensing steam to develop the correlation parameters. The pressure drop is calculated on an incremental basis and summed over the condenser length. The results of this method compare favorably with empirical measurements.

Abst

CONDENSING PRESSURE DROP BY IMPROVED LOCKHART-MARTINELLI CORRELATION

by F. L. Robson and W. E. Hilding

Mechanical Engineering Department, University of Connecticut

SUMMARY

The object of the work described below was to develop a method for predicting the pressure loss of condensing flow. By using a correlation similar to that of Lockhart and Martinelli, a procedure was developed which allowed the successful prediction of incremental and overall pressure drops. This correlation is based upon actual condensing flow rather than two-phase, two-component flow. While the flow rates were in the range from 40 lbm/hr to 500 lbm/hr in 1/4 inch and 1/2 inch tubes, it can be concluded that the results are applicable to other flow rates and pipe diameters so long as the Reynolds number lies in the range of the tests, i.e., 50,000 to 200,000.

Nomenclature

A	- area - ft^2
C_p	- specific heat - BTU/lbm
D	- tube diameter - ft
f	- Fanning friction factor
G	- mass velocity - lbm/sec ft^2
g	- gravitational acceleration - 32.17 - ft lbm/sec^{2}lbf
J	- mechanical equivalent of heat - 778 ft lbf/BTU
K	- constant in Eq. 5 friction factor equation - 0.046 turbulent - 16 laminar
L	- tube length - ft

Nomenclature (cont.)

P	- pressure - lbf/ft^2
n	- number of increment
$(\Delta P/\Delta Z)_v$	- vapor phase pressure drop gradient - lbf/ft^2/ft
$(\Delta P/\Delta Z)_l$	- liquid phase pressure drop gradient - lbf/ft^2/ft
$(\Delta P/\Delta Z)_{tp}$	- two-phase friction pressure drop gradient - lbf/ft^2/ft
$(\Delta P/\Delta Z)_m$	- momentum pressure drop gradient - lbf/ft^2/ft
$(\Delta P/\Delta Z)_T$	- total pressure drop gradient - lbf/ft^2/ft
Q	- heat flux - BTU/sec ft
Re	- Reynolds number
T	- temperature - ^oR
V	- velocity - ft/sec
W	- mass flow rate - lbm/sec
X	- Lockhart-Martinelli parameter
z	- length along condenser - ft
ρ	- density - lbm/ft^3
μ	- viscosity - lbm/sec ft
λ	- heat of vaporization - BTU/lbm
ϕ	- ratio of pressure at point in tube to entrance pressure
ϕ_g	- Lockhart-Martinelli two-phase correlation factor

Subscripts:

e	- entrance condition	vt	- viscous liquid - turbulent vapor
l	- liquid	tt	- turbulent liquid - turbulent vapor
v	- vapor	T	- total

Superscript:

c - exponent of variation λ with P

INTRODUCTION

The projected use of nuclear-Rankine cycle powerplants for space applications has intensified the interest in compact, lightweight heat exchangers. The condenser, either of the shell and tube variety or a condensing-radiator, can represent a sizable percentage of powerplant weight. Work done on condenser optimization (1), (2)* has shown that the condensing pressure drop is a major criterion for optimization. A definite need now exists for an accurate method of predicting two-phase pressure losses in condensing flow.

Martinelli et al (3), Lockhart and Martinelli (4), and Martinelli and Nelson (5) were among the first to propose a method applicable to condensation in tubes. Since the appearance of these papers many other investigators have published so-called Lockhart-Martinelli type correlations for pressure losses in two-phase systems. Of these, the paper of Baroczy and Sanders (6) proposes a method to be used for pressure drop predictions for a direct-radiator condenser with a constant heat transfer rate. Like the previous work, however, this method is based on two-component (nitrogen-mercury), two-phase correlations. The work in (6) was applied to the data from tests at the University of Connecticut (7), (8) but the results were not in good agreement with the experiments. It was then decided to modify the approaches of (4) and (6) using data for condensing steam. The results of this analysis agree well with the experimental data.

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* Numbers in parentheses designate References at end of paper.

METHOD OF ANALYSIS

The condensing system under consideration consists of a long, small diameter, straight tube with an annular cooling water jacket. The tubes have inside diameters of 0.190 inches and 0.550 inches with condensing lengths varying from 4 to 11 feet. The flow rates range from 40 lbm/hr to nearly 500 lbm/hr.

No assumption is made as to the flow mechanism, i.e., annular, annular mist, fog, etc., although the data used are from tests run in the annular and annular mist regions (9). The only assumptions are that the vapor phase can be considered saturated at all times and that the static pressure is uniform at any cross section of the tube.

The papers of Baroczy and Sanders (6) and Lockhart and Martinelli (4) present the basic theory of two-phase pressure drop. The static pressure drop in a condensing system is postulated to consist of two different pressure changes: $(\Delta P)_m$ the momentum change and $(\Delta P)_{tp}$ the friction change. The total is then the algebraic sum of these. $(\Delta P)_m$ and $(\Delta P)_{tp}$ are calculated on an incremental basis and the overall pressure loss is then

$$(\Delta P)_T = \sum \left[(\Delta P/\Delta Z)_m + (\Delta P/\Delta Z)_{tp}\right]\Delta Z \qquad \text{Eq. 1}$$

The friction loss $(\Delta P)_{tp}$ is found by means of the Lockhart-Martinelli correlation. A factor ϕ_g is found as a function of X, the Lockhart-Martinelli parameter. This parameter is the ratio of the pressure drop of the liquid flowing alone in the tube, to the pressure drop of the vapor flowing alone. Previous treatments have assumed that the vapor properties can be treated as constants, the values being taken, for example, as those occurring at the average static pressure. If the pressure drop is more than 20 percent of the initial pressure, this assumption can lead to appreciable errors. A method has been developed (10) which allows the evaluation of fluid properties on each increment of condenser length.

If the property under consideration, for instance the density, were plotted versus saturation pressure, the resulting curve on a log-log plot (Fig. 1) would allow the formulation of the following equation:

$$\rho_1/\rho_2 = (P_1/P_2)^j \qquad \text{Eq. 2}$$

If the denominator is then the condition at tube entrance, (which is known) the density at the entrance of any increment is then

$$\rho = \rho_e(P/P_e)^j \qquad \text{Eq. 3}$$

The value of P is known since the static condensing loss has been calculated for all the preceding increment(s). That this method can be used to evaluate the properties over a range of saturation pressures which is likely to be encountered in many applications is shown in Figs. 2 and 3. These figures show the small variation of the exponent with pressure.

The vapor phase pressure drop is given by the Fanning equation

$$(\Delta P)_v = 2fG_v{}^2\Delta Z/Dg\rho_v \qquad \text{Eq. 4}$$

where

$$G_v = W_v/A$$

and, for turbulent vapor flow $f = 0.046/(Re)_v{}^{0.2}$.

Based on the "superficial"[1] Reynolds number, the vapor flow was entirely in the turbulent region ($Re > 2000$), while the liquid "superficial" Reynolds number changed from viscous ($Re < 2000$) to turbulent.

The Lockhart-Martinelli parameter is then, for the viscous liquid-turbulent vapor region

$$X_{vt} = \left[K_l/K_v \, \rho_v/\rho_l \, \mu_l/\mu_v \, W_l/W_v \right]^{0.5} / (Re)_v^{0.4} \qquad \text{Eq. 5}$$

and in the turbulent liquid-turbulent vapor region

$$X_{tt} = \left[(W_l/W_v)^{1.8} \, (\rho_v/\rho_l) \, (\mu_l/\mu_v)^{0.2} \right]^{0.5} \qquad \text{Eq. 6}$$

In order to evaluate X's and the pressure drops, the vapor and liquid flow rates must be known. If the heat flux is constant along the length of the tube, a simple linear relation can be established (Ref. 6).

$$W_v = W_e(1 - Z/L) \qquad \text{Eq. 7}$$

However, if there is an arbitrary heat flux, the incremental flux must be determined. Any of the standard heat transfer texts (11), (12), etc., outline methods for flux determinations for shell and tube heat exchangers. Having established a value for Q, an approximate value for the amount of vapor condensed, (the increase of the liquid flow in the increment) is given by

$$W_l = Q/\lambda \qquad \text{Eq. 8}$$

---

[1] The "superficial" Reynolds number is based on the flow being only vapor or only liquid. Actually, the flow area available to either phase is less than the tube cross-sectional area, although the sum of flow areas equals that of the tube.

In condensing, the momentum pressure drop is negative, that is, it lessens the effective static pressure drop. The value of the momentum change, neglecting the liquid because of its relatively small velocity, is

$$(\Delta P)_m = -\rho_v V_v^2/2g \qquad \text{Eq. 9}$$

In the determination of the two-phase pressure drop, an iterative method was used for a more exact evaluation of the liquid flow rate. The entire process was programmed for solution on a digital computer.

To determine a more exact condensing rate, an energy balance on the increment is solved for W_L (neglecting liquid kinetic energy and vapor momentum change in the increment)

$$Q = W_1(\lambda + V_v^2/2gJ) + W_v C_{pv} \Delta T_v + W_1 C_{pl} \Delta T_1 \qquad \text{Eq. 10}$$

where the value of dT is found from an exponential relationship with pressure and

$$T = T_e\, \phi^e \qquad \text{Eq. 11}$$

$$\Delta T = T_e - T \qquad \text{Eq. 12}$$

Thus it seems that the value of $(\Delta P)_T$ must be known before we can solve for it. The iterative process, however, goes through the pressure drop calculations again and again until two consecutive values of W_1 agree to within $\pm$ 0.05 percent. The value of $(\Delta P)_T$ found this way is then subtracted from the pressure at condenser tube entrance and this is then the pressure at the beginning of the second increment.

The vapor velocity appears in the equation for momentum pressure drop and in the energy balance equation. In order to determine V_v, the fraction of the tube filled with vapor must be known. Several methods have been proposed to find this value (13), (14) but it was the curve of Lockhart-Martinelli (3) that was used. V_v is then found from the continuity equation.

The determination of the Lockhart-Martinelli two-phase pressure drop factor "ϕ_g", was the reverse of the preceding calculations. Since the actual pressure drops and heat transfer rates were known, the flow variables and condensing rates could be calculated, gas phase and momentum pressure drops can be found, and the value of ϕ_g will be given by

$$\phi_g = \left[\frac{(\Delta P)_T - (\Delta P)_m}{(\Delta P)_v} \right]^{0.5} \qquad \text{Eq. 13}$$

X_{vt} or X_{tt} was calculated and ϕ_g was plotted as a function of X and Reynolds number (Fig. 4). The values found in this manner fall above the Lockhart-Martinelli curve, approaching it only at low (less than 50,000) Reynolds numbers. The values proposed by Baroczy and Sanders (6) show better agreement.

RESULTS

Figs. 5 and 6 show the comparison of experimental and calculated results for the two tests having the highest and lowest mass flow rates. One of these tests demonstrates the fact that a local rise in static pressure is possible because of momentum pressure regain. A comparison of pressure drops as predicted by the Lockhart-Martinelli (4) method and the method of this paper is shown in Fig. 7.

Part of the differences between experimental and predicted values can be attributed to difficulty in accessing the correct value of void fraction.

APPENDIX

The calculation of the incremental pressure drop was done in the following manner:

The initial estimate of liquid flow rate

$$W_l = Q/\lambda$$

The vapor flow rate

$$W_v = W_e - W_l$$

Vapor pressure drop

$$(\Delta P/\Delta Z)_v = (0.046)W_v^{\,2}/(0.785)^2 D^5 g\, \rho_v (Re)_v^{\,0.2}$$

Momentum pressure drop

$$(\Delta P/\Delta Z)_m = -\rho_v V_v^{\,2}/2g$$

The parameter X is calculated (for viscous-liquid turbulent vapor)

$$X_{vt} = \left[(16/0.046)(\rho_v/\rho_l)(\mu_l/\mu_v)(W_l/W_v)\right]^{0.5}/(Re)_v^{\,0.4}$$

and a value of ϕ_g found from the plot of ϕ_g versus X.

The two-phase friction drop is then

$$(\Delta P/\Delta Z)_{tp} = \phi_g^{\,2}\,(\Delta P/\Delta Z)_v$$

The total pressure drop is then

$$(\Delta P/\Delta Z)_T = (\Delta P/\Delta Z)_{tp} + (\Delta P/\Delta Z)_m$$

Now, having determined an initial $(\Delta P)_T$ for the increment, the pressure dependent variables are found

$$\lambda = \lambda_e/\phi^c$$

$$T = T_e\phi^e$$

$$\mu_v = \mu_{ve}\phi^i$$

$$\rho_v = \rho_{ve}\phi^j$$

A new value of W_l is found

$$W_l = (Q - W_v C_{pv} \Delta T_v - W_l C_{pl} \Delta T_l)/(\lambda_e \phi^c + V_v{}^2/2gJ)$$

Using this value of W_l a new vapor flow is calculated and the vapor velocity is found

$$V_v = W_v/\rho_v A_v$$

The new values of W_v and V_v and used to recalculate the friction and momentum drops, X_{vt}, etc. The process is repeated until two consecutive values of W_l agree to within $\pm$ 0.05 percent.

When the above agreement is reached, the value of the $(\Delta P)_T$ is subtracted from the entrance pressure. The results of the first increment are used as the entrance conditions for the second increment. When correctly programmed, a tube of twenty increments can be solved for in about 20 minutes on an IBM 1620. Table 1 shows numerical results for one of the tests.

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TABLE I. SAMPLE CALCULATIONS FOR TEST 8-14-52. TOTAL CONDENSING LENGTH FIVE FEET.

INITIAL CONDITIONS

$P_e = 4210$ LBF/FT^2 $\quad V_{ve} = 826$ FT/SEC $\quad \rho_V = 0.071$ LBM/FT^3

$W_e = 0.0104$ LBM/SEC $\quad D_t = 0.19$ IN $\quad \rho_L = 58.8$ LBM/FT^3

Z	Q	W_L	W_V	Re_v	$\left(\frac{\Delta P}{\Delta Z}\right)_v$	X	Φ_g	$\left(\frac{\Delta P}{\Delta Z}\right)_{tp}$	V_v	$\left(\frac{\Delta P}{\Delta Z}\right)_M$	ΔP_T	P
	$\times 10^3$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^4$	$\times 10^2$	$\times 10^{-2}$		$\times 10^2$	$\times 10^2$	$\times 10^2$	$\times 10^2$	
0.5	4.48	1.3	9.3	7.96	7.01	1.05	1.36	12.9	8.65	8.5	4.4	28.9
1.0	9.2	2.63	8.17	7.06	6.04	1.61	1.45	12.7	8.90	8.7	4.0	23.4
1.5	13.5	3.95	7.14	6.28	5.30	2.08	1.5	10.7	8.85	7.2	3.5	20.9
2.0	17.0	4.83	6.37	5.68	4.69	2.45	1.55	11.1	8.72	8.2	2.9	18.9
2.5	22.1	6.31	5.08	4.60	3.94	3.26	1.58	9.8	8.05	7.7	2.12	17.4
3.0	26.2	7.49	4.10	3.76	2.93	4.13	1.57	7.3	7.15	5.8	1.55	16.3
3.5	30.1	8.62	3.17	2.99	2.10	8.53	1.58	5.3	6.08	4.3	1.0	15.7

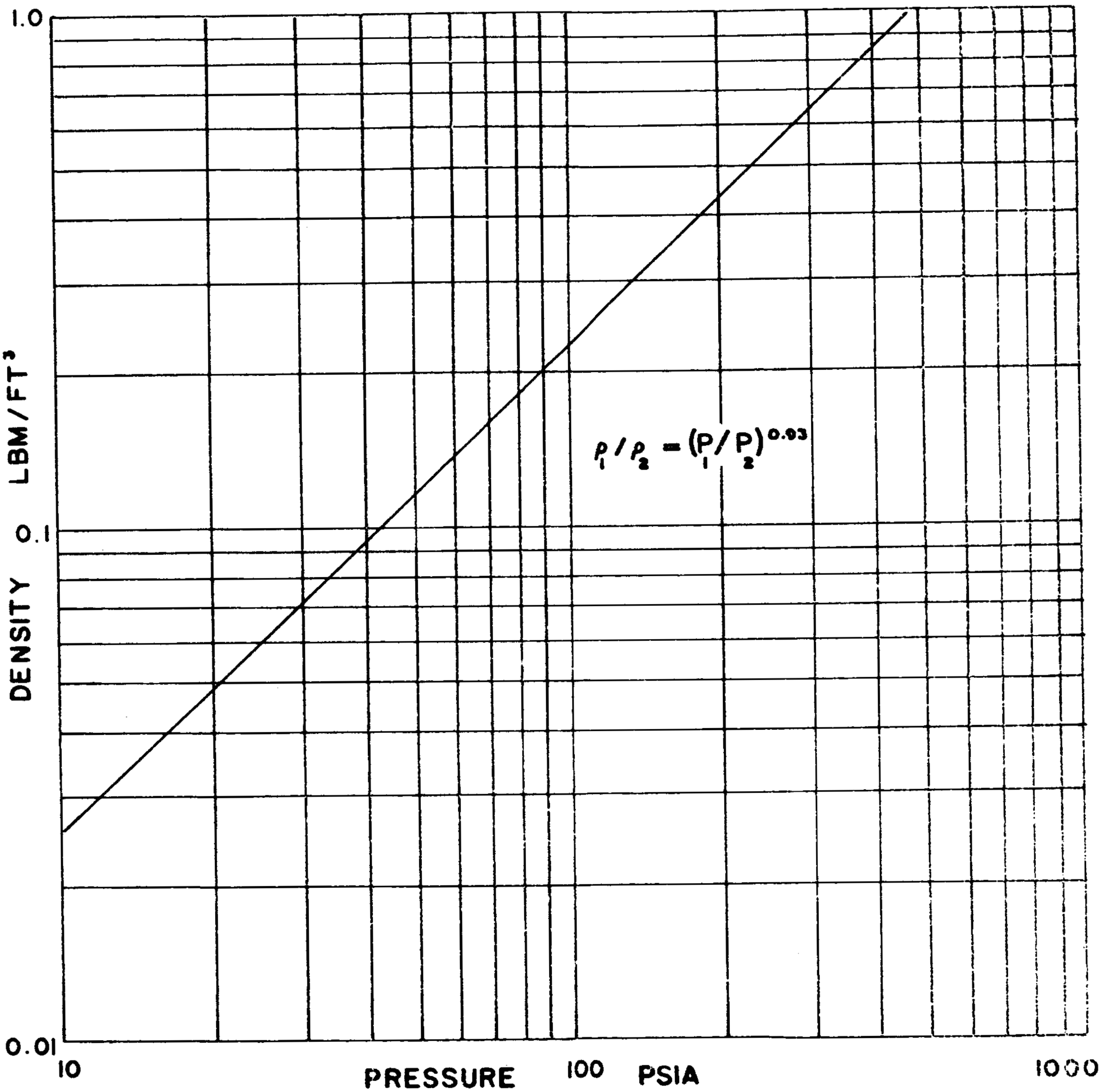


FIG I RELATION BETWEEN DENSITY AND SATURATION PRESSURE

FIG 2

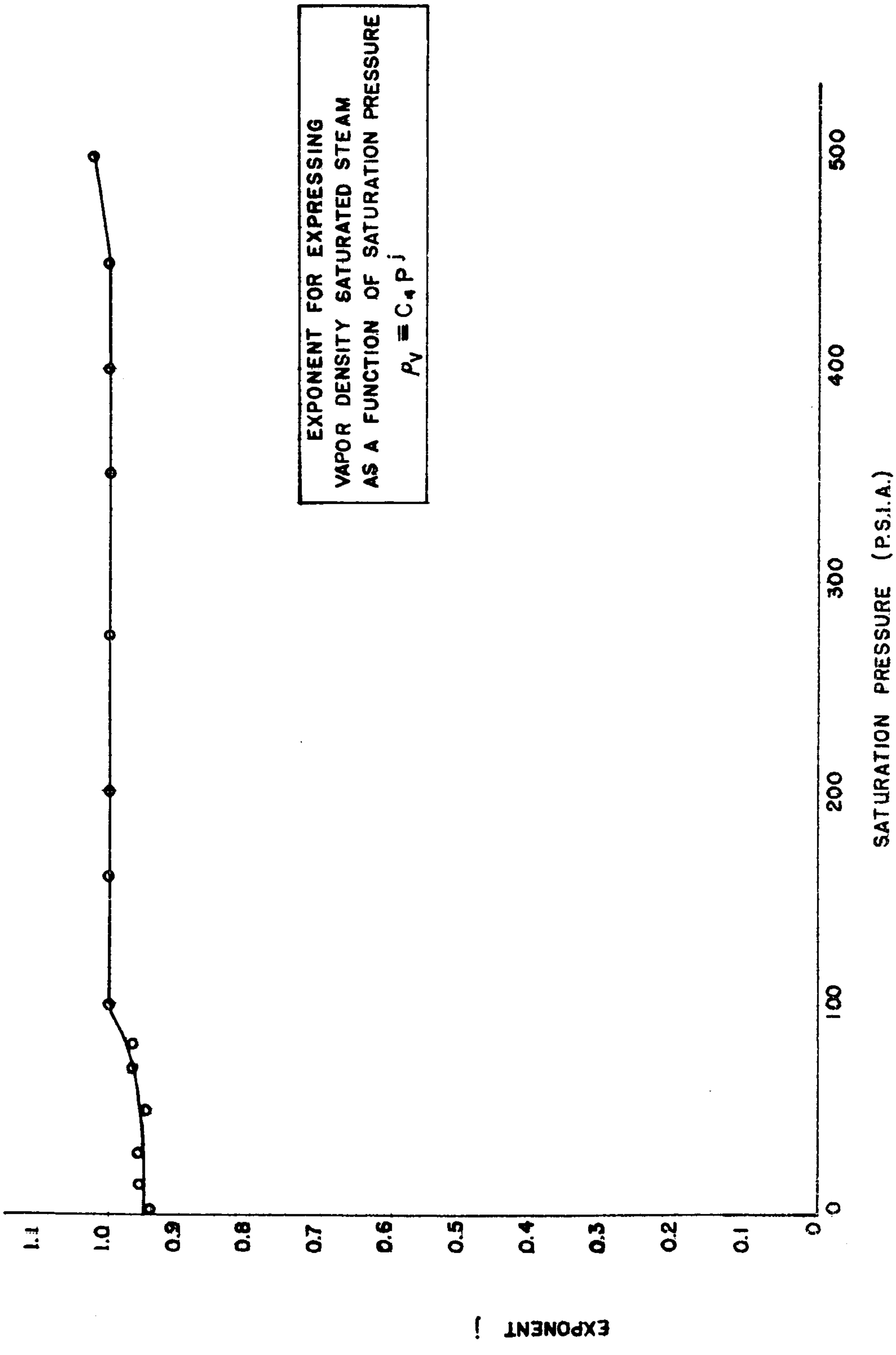
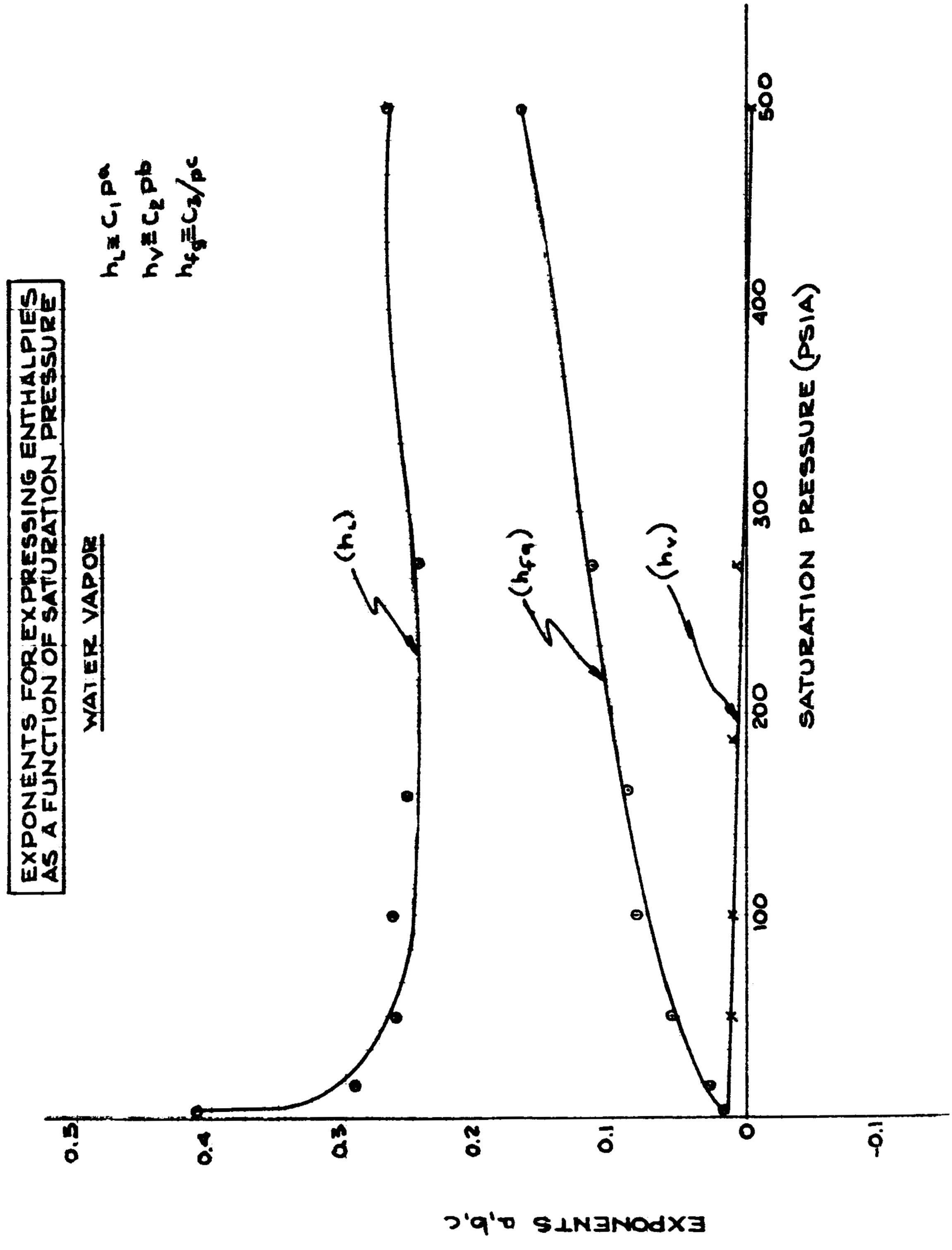

EXPONENT FOR EXPRESSING
VAPOR DENSITY SATURATED STEAM
AS A FUNCTION OF SATURATION PRESSURE
$\rho_V \equiv C_4 P^j$
SATURATION PRESSURE (P.S.I.A.)
0
100
200
300
400
500
EXPONENT j
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1

FIG 3



EXPONENTS FOR EXPRESSING ENTHALPIES AS A FUNCTION OF SATURATION PRESSURE
WATER VAPOR
$h_L \equiv C_1 p^a$
$h_V \equiv C_2 p^b$
$h_{fg} \equiv C_3 / p^c$
(h_L)
(h_{fg})
(h_V)
0.5
0.4
0.3
0.2
0.1
0
-0.1
EXPONENTS a,b,c
100
200
300
400
500
SATURATION PRESSURE (PSIA)

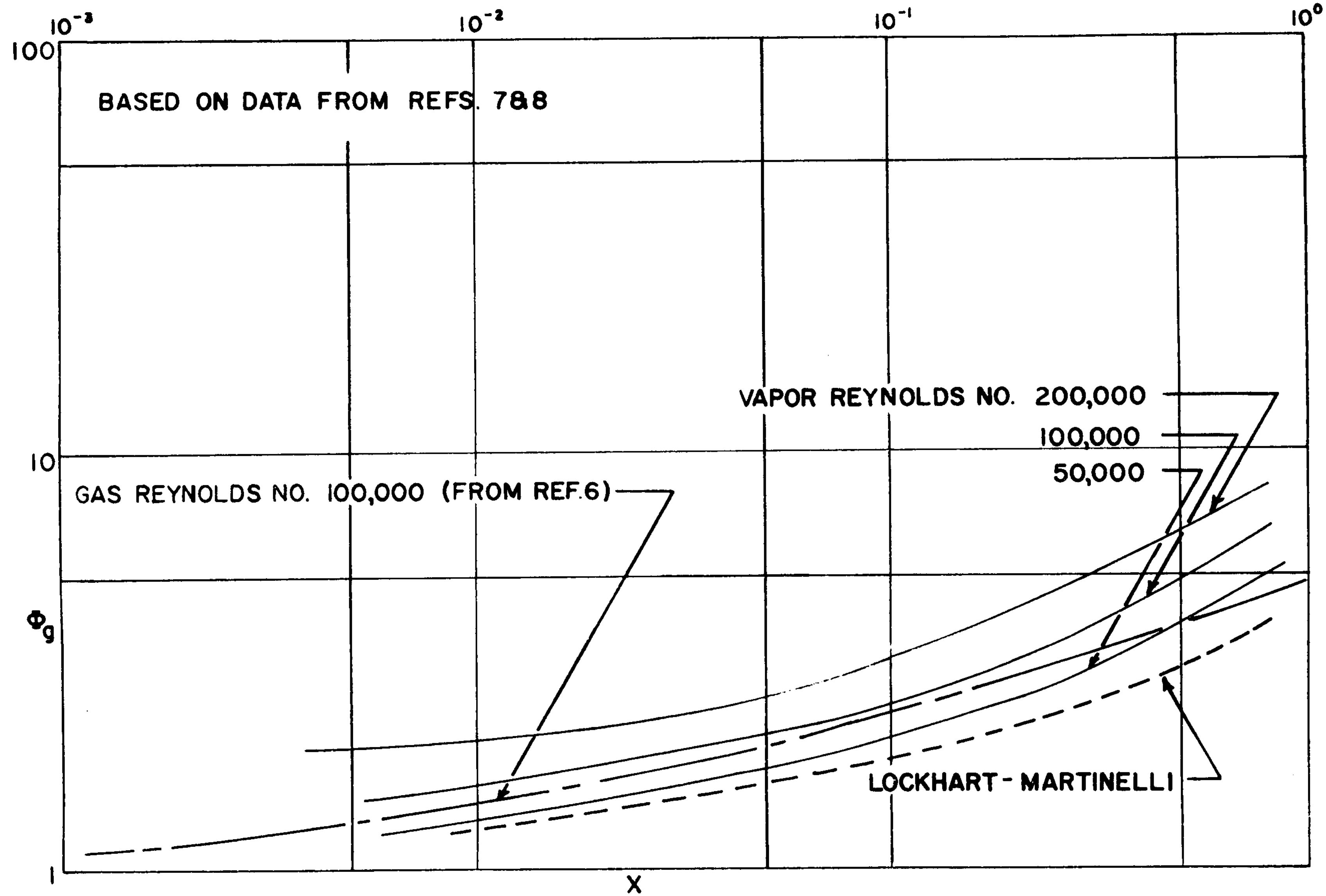


FIG 4 CORRELATION OF LOCKHART-MARTINELLI PARAMETER FOR STEAM

FIG 5

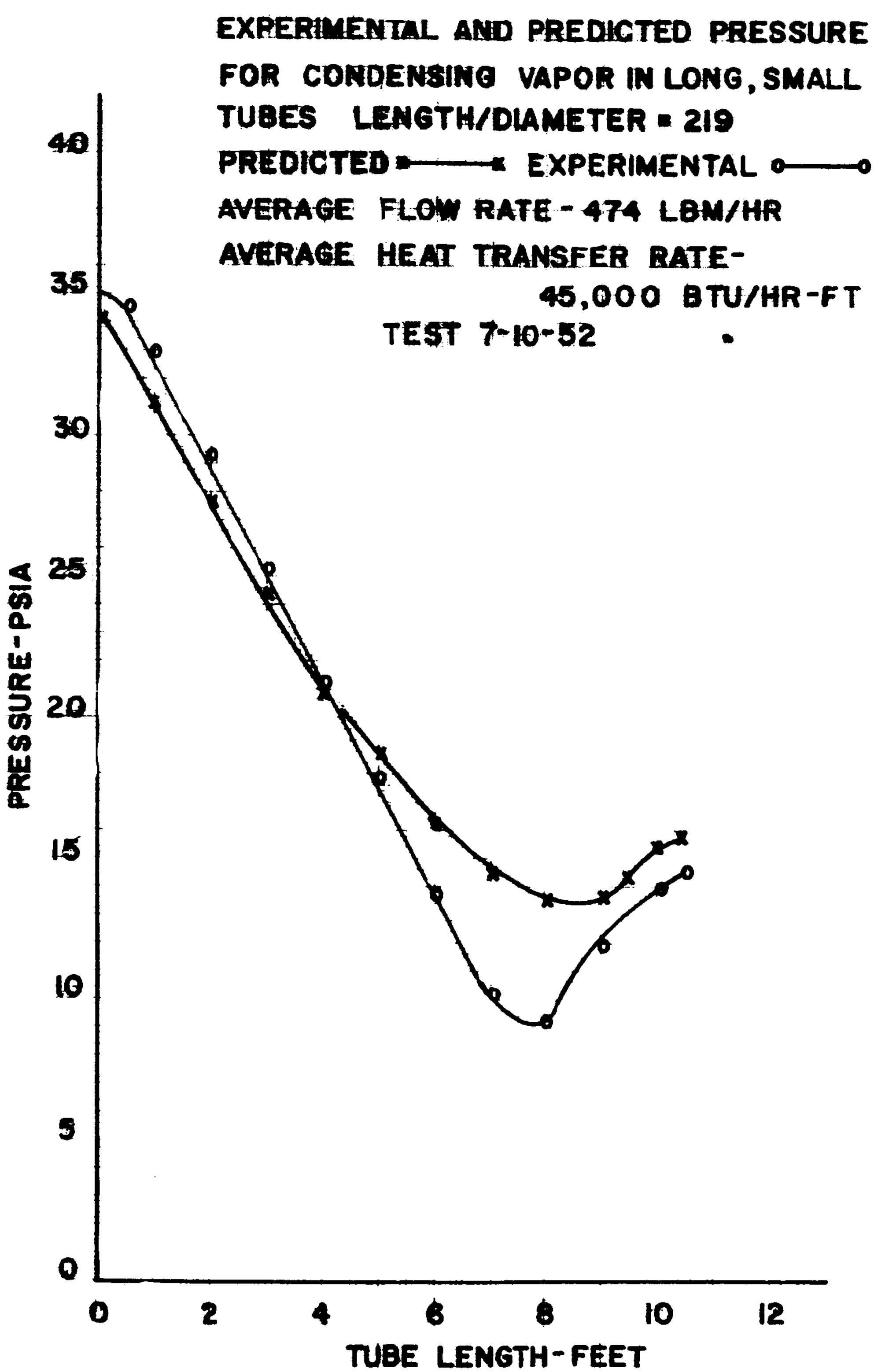
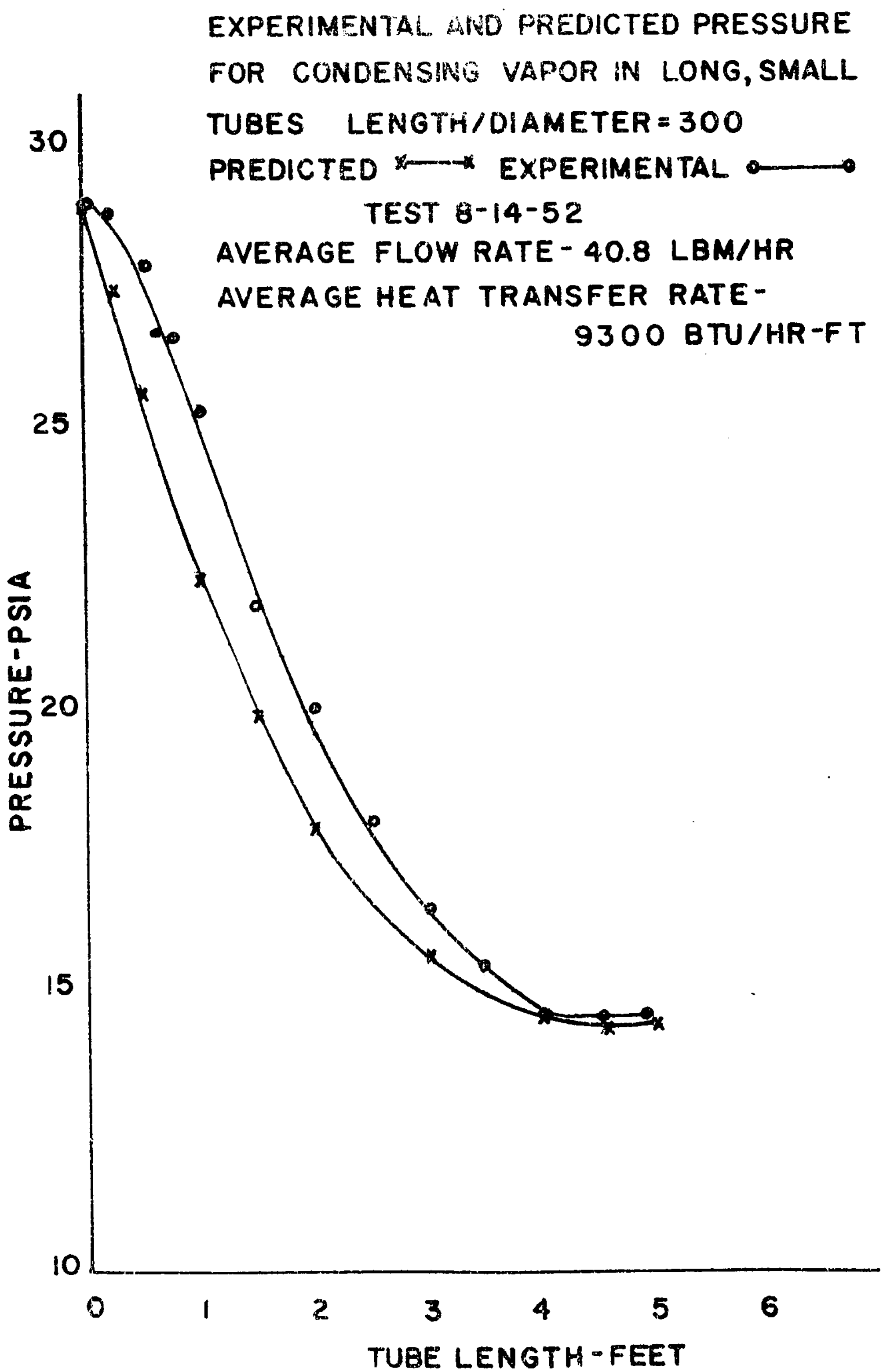
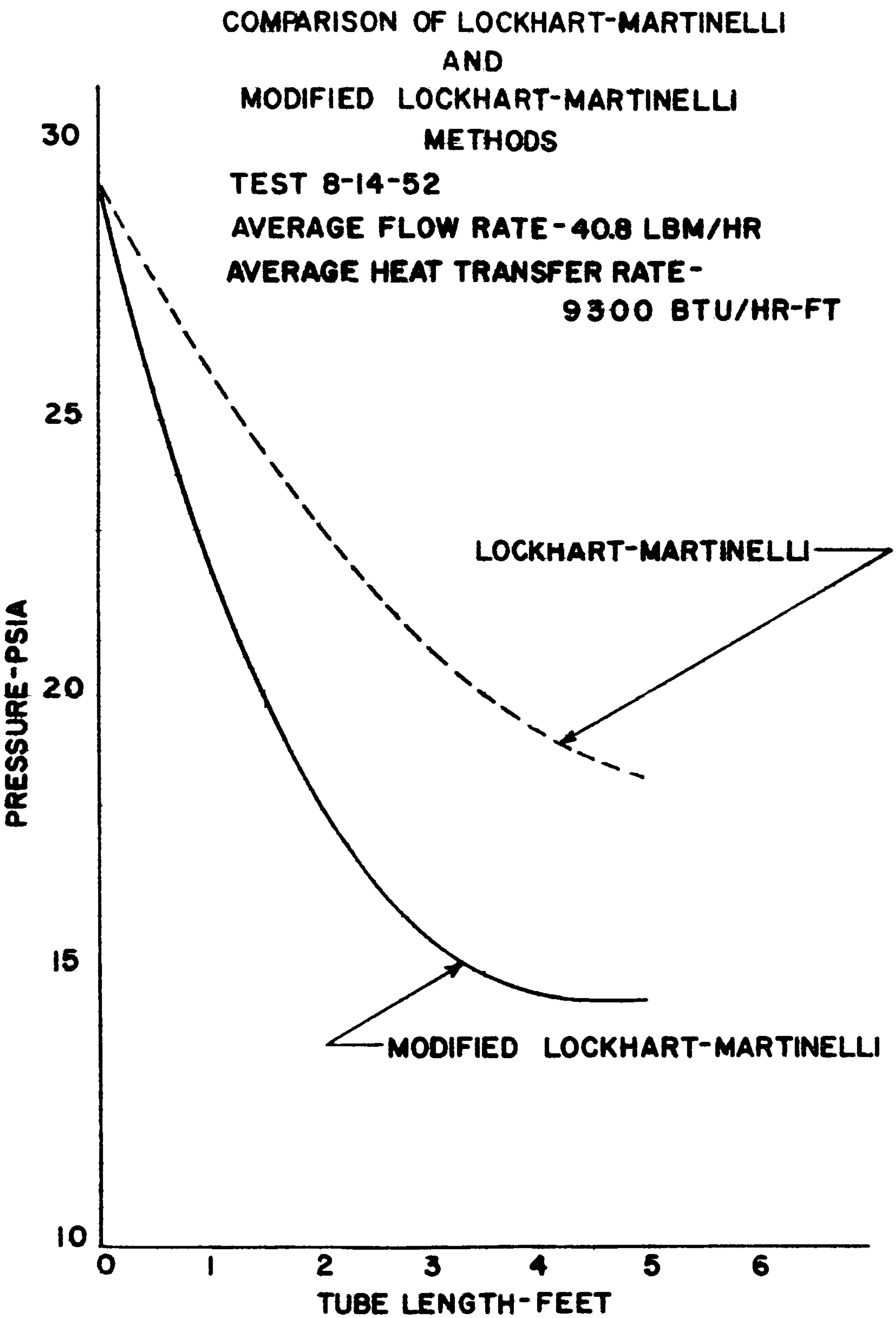

EXPERIMENTAL AND PREDICTED PRESSURE
FOR CONDENSING VAPOR IN LONG, SMALL
TUBES LENGTH/DIAMETER = 219
PREDICTED x——x EXPERIMENTAL o——o
AVERAGE FLOW RATE - 474 LBM/HR
AVERAGE HEAT TRANSFER RATE-
45,000 BTU/HR-FT
TEST 7-10-52
PRESSURE - PSIA
40
35
30
25
20
15
10
5
0
0
2
4
6
8
10
12
TUBE LENGTH - FEET

FIG 6



EXPERIMENTAL AND PREDICTED PRESSURE FOR CONDENSING VAPOR IN LONG, SMALL TUBES LENGTH/DIAMETER = 300
PREDICTED
EXPERIMENTAL
TEST 8-14-52
AVERAGE FLOW RATE - 40.8 LBM/HR
AVERAGE HEAT TRANSFER RATE - 9300 BTU/HR-FT
PRESSURE-PSIA
30
25
20
15
10
0
1
2
3
4
5
6
TUBE LENGTH-FEET

FIG 7


COMPARISON OF LOCKHART-MARTINELLI
AND
MODIFIED LOCKHART-MARTINELLI
METHODS
TEST 8-14-52
AVERAGE FLOW RATE-40.8 LBM/HR
AVERAGE HEAT TRANSFER RATE-
9300 BTU/HR-FT
LOCKHART-MARTINELLI
MODIFIED LOCKHART-MARTINELLI
PRESSURE-PSIA
30
25
20
15
10
0
1
2
3
4
5
6
TUBE LENGTH-FEET